DEPARTMENT OF THE NAVY
SCIENTIFIC AND TECHNICAL INTELLIGENCE CENTER
TRANSLATION DIVISION
4301 SUITLAND ROAD,
SUITLAND, MARYLAND 20390

CLASSIFICATION: UNCLASSIFIED

APPROVED FOR PUBLIC RELEASE, DISTRIBUTION UNLIMITED

TITLE: Reduction of the Noise Level in Centrifugal Fans by Means of Transition Meshes

Snizheniye vozdushnogo shuma v tsentrobezhnykh ventilyatorakh setchatymi turbulizatorami

AUTHOR(S): Petrov, Yu. I., Khoroshev, G. A., and Novozhilov, S. Ya.

PAGES: 8

SOURCES: Sudostroyeniye No. 5, 1970
Pages 22-25

ORIGINAL LANGUAGE: Russian

TRANSLATOR: C

NAVSTIC TRANSLATION NO. 3322

APPROVED P T K

DATE 30 June 1972

REDUCTION OF THE NOISE LEVEL IN CENTRIFUGAL FANS BY MEANS OF TRANSITION MESHES

Yu. I. Petrov, G. A. Khoroshev, S. Ya. Novozhilov

The spectral composition of the noise from centrifugal fans contains intensive components precisely in that part of the spectrum where the human ear has the highest sensitivity (500 - 2000 Hz). For this reason, the use of centrifugal fans in rooms with personnel necessitates the installation of fairly large mufflers. However, in those cases when the use of mufflers is impossible or limited due to other considerations (for example, restrictions on the weight and size), one has to take measures that will reduce the noise level at the source itself, i.e., in the fan.

The study [1] shows that, in a number of cases, the fan noise can be reduced by selecting an optimum value for their pressure factor. Here the dimensions and weight of fans may remain the same, but the noise generated will be reduced by 5 - 8 dB as compared with fans designed according to normal methods [2].

As shown in studies, a further reduction of the fan noise can be achieved only by changing the structure of the flow through the fan ducts. The basic idea behind the reduction of noise by means of transition meshes is that we want to create conditions in the interblade fan ducts such that the formation of large vortices will be prevented and the vortices themselves will be broken up into smaller ones. As a result, the noise spectrum of a fan with transition meshes becomes somewhat modified. The low frequency components of the noise spectrum (from 50 to 2000 Hz) are substantially reduced,

whereas the high-frequency components become somewhat intensified.

The method proposed in [3] for reducing noise with the aid of transition meshes may be used primarily in fans with a separated flow around the rotor blades.

In this case a transition mesh, placed near the leading edges, changes the velocity distribution along the blade profile, and — by turbulizing the boundary layer — it displaces the point of flow separation on the profile toward the trailing blade edges, which increases the angle of turn of the flow in the lattice, i.e., it increases the angle of turn of the flow in the lattice, i.e., it increases the profile lift factor C_y. Transition meshes are also conducive to reducing the vortex dimensions. This is also manifested in the fact that small vortices, which form behind the mesh and having a high energy, break up the larger vortices. This is accompanied by a sharp increase in the noise frequency characteristic of small vortices, and consequently the noise spectrum shifts into the region of higher frequencies.

The use of transition meshes at the same time results in a certain increase of the drag of the fan as a whole and, as a result, a reduction of the pressure produced by the fan. In order to compensate for this factor, it is necessary to increase the fan rpm, which leads to an increase in its noise level. However, as shown in several studies, the gain due to the reduction of the total fan noise as a result of a displacement of its spectrum toward the high-frequency region is in a majority of cases more substantial, even with a simultaneous increase in the rpm in order to obtain a given pressure.

In the case of unseparated flow around the wheel blades, the acoustic effectiveness of transition meshes is manifested in the form of a reduction of the vortex size, i.e., in an increase of the noise frequency. In such fans, one usually observes a greater reduction of the lift factor for the profile lattice, C_y, and the increase of the flow speed necessary to compensate for the pressure drop sometimes nullifies any gains obtained by using transition meshes.

Recommendations as to the selection of transition meshes. At the present time, no data are available that would permit a selection of the transition mesh parameters (active cross section and wire dimensions) and their location when the fan is designed. However, certain general recommendations can already be formulated. First of all, it is necessary that the power of the eddy stream behind the transition mesh be sufficient for a substantial increase in the energy flow to the boundary layer and for a displacement of the flow separation point toward the trailing edges. The energy of the vortices must also be sufficient for the disintegration of large vortices in ducts, and their size must correspond to frequencies not exceeding 2 - 4 kHz.

All these conditions can be achieved by a proper choice of the transition mesh parameters and the location where they are installed. Figure 1 shows the results of measuring the spectral composition of the air noise for a mass-produced ship centrifugal fan with two types of transition meshes. As we can see from the plots, the smaller mesh (1.3 x 1.3 x 0.25 mm) reduces the noise in the 100 - 10,000 Hz range by 4 - 9 dB. At high frequencies (above 10,000 Hz) its acoustic efficiency decreases. The larger transition mesh (5.4 x 4.5 x 1.4 mm) sharply reduces the noise in the 100 - 2000 Hz range. However, at frequencies above 2000 Hz, the noise level is higher than for the initial fan. Thus, the selection of the transition mesh size must be based on the spectral region in which the noise level must be lowered. If the most intensive components of the vortex hoise of the fan are concentrated in the region up to 1000 Hz, then one can use transition meshes with larger wire diameters. If, however, the noise is concentrated in the 1000 - 2000 Hz range, then smaller wire diameters must be used (assuming that the active cross section of the mesh is equal to approximately 0.6). It must always be kept in mind that the mesh with larger wire diameters will simultaneously result in an increase of the noise level in the lower frequency region. In any case, the mesh size can be only selected experimentally at the present time.

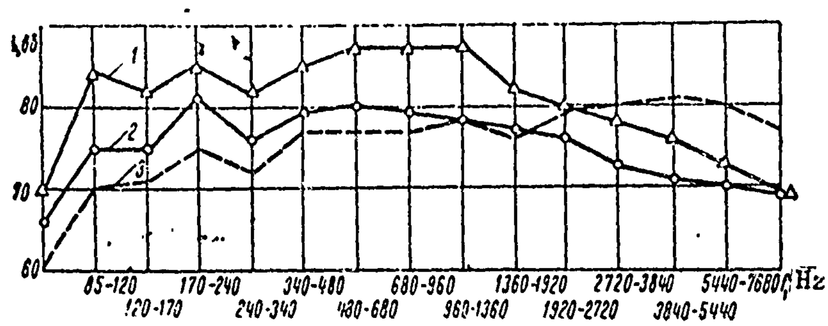

Figure 1. Effect of mesh parameters on the noise generated by a centrifugal fan (outer diameter of rotor wheel is D_2 = 3.25 mm; n = 2870 rpm; z = 24 blades).
1 - Initial fan (Q = 4100 m^3/hr; H = 146 mm of water; η = 0.626); 2 - the same fan but with a transition mesh (1.3 x 1.3 x 0.25 mm) on the leading and trailing edges of the rotor blades (Q = 4100 m^3/hr; H = 138 mm of water; η = 0.545); 3 - the same fan but with a transition mesh (5.4 x 5.4 x 1.4 mm) on leading and trailing edges of rotor blades (Q = 4100 m^3/hr; H = 135 mm of water; η = 0.536).

At the present time, it is recommended that transition meshes be placed directly on the leading or trailing fan edges (Figure 2). In the first case, the acoustic efficiency of the transition mesh will be much greater, and will amount approximately to 70 - 80% of the total noise reduction level.

As shown in many studies of fans with transition meshes, their acoustic efficiency increases as the flow conditions around the fan profile lattice become worse. Figures 1 and 3 show the results of tests made on two centrifugal fans. In the first case, a transition mesh results in a noise level reduction by 10 - 13 dB, whereas in the second case no noticeable reduction of the noise level is observed. The different effect of the transition meshes is due to the fact that, in the second case, the local velocity increase factor on the inoperative side of the blades is 1.8 in relation to

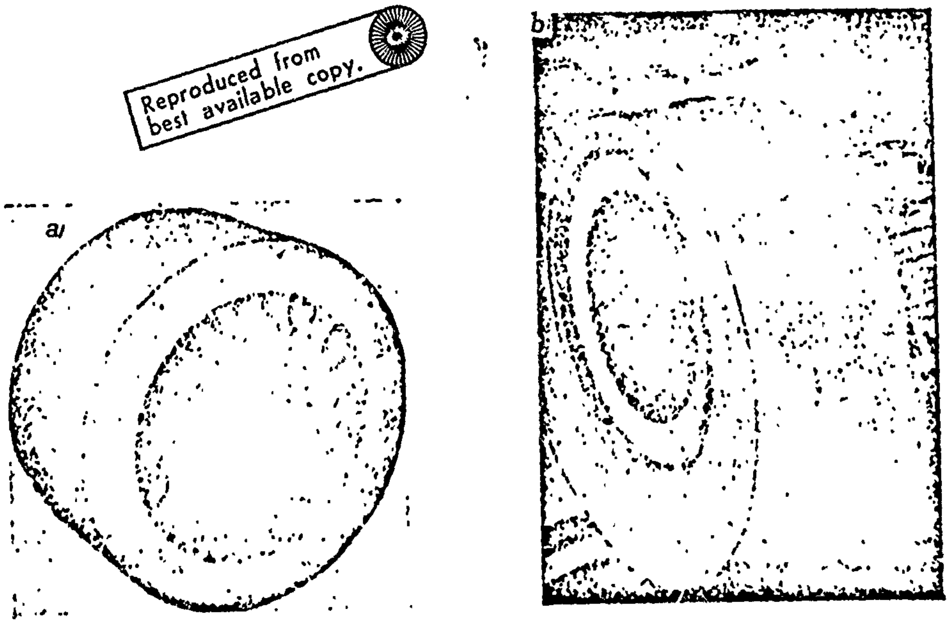

Figure 2. General view of a rotor with a mesh on the leading and trailing blade edges (a) and a view of the fan housing (b).

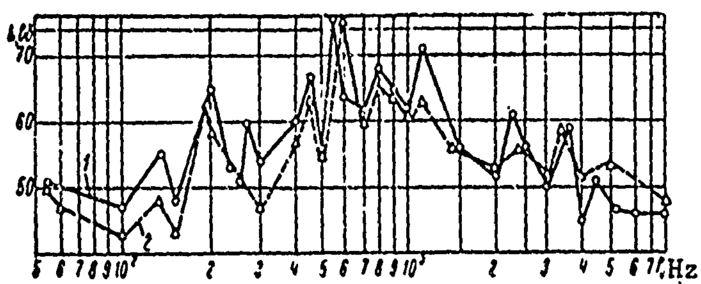

Figure 3. Air noise spectrum for a centrifugal fan (Q = 300 m^3/hr; H = 165 mm of water; D_2 = 300 mm; n = 2870 rpm; z = 12 blades).

1 - Initial fan; 2 - fan with a mesh on the leading and trailing edges of rotor blades.

the flow velocity (Figure 4), whereas in the first case it is equal to 5. The latter indicates that the flow conditions around the profile lattice in the first fan are much worse than in the second.

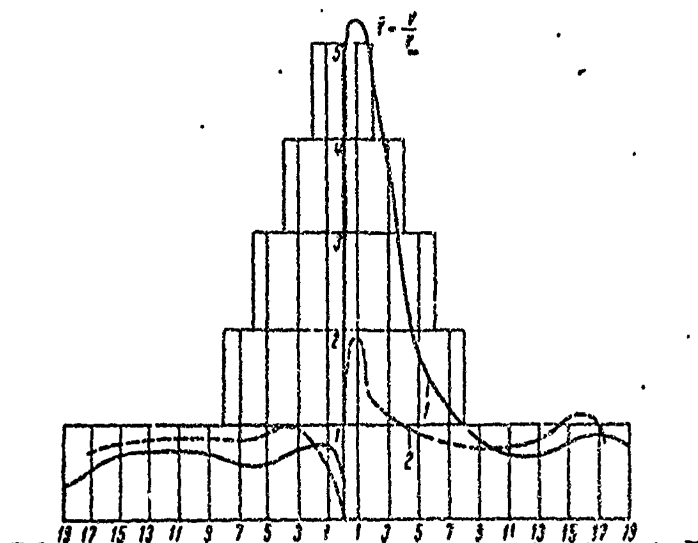

Figure 4. Local velocity factor along the profile of centrifugal fan blades.

1 - Fan with Q = 4000 m^3/hr; H = 165 mm of water; η = 0.68; D_2 = 325 mm; 2 - fan with Q = 300 m /hr; H = 165 mm of water; η = 0.7; D_2 = 300 mm.

The use of transition meshes in fans with unsatisfactory flow conditions around blades may result in an improvement of the energy characteristics of the fan as a whole. Figure 5 shows the noise spectra for a fan in which the installation of a transition mesh reduced the noise level by 5 - 12 dB and increased the pressure by 3 mm of water, the efficiency being constant. However, in a majority of cases transition meshes somewhat worsen the energy characteristics of fans, where efficiency is reduced by 3 - 8%, and pressure by 5 - 15 mm of water. In spite of this fact, transition meshes can still be recommended, since the acoustic effect associated with their use cannot

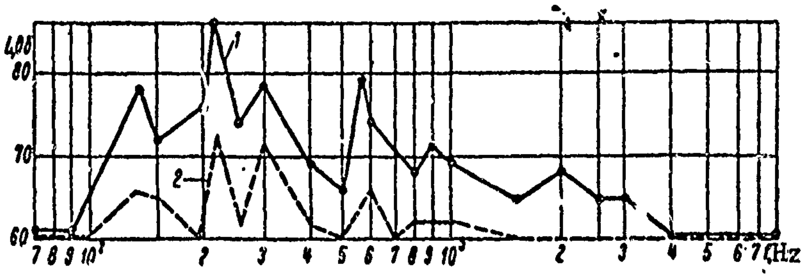

Figure 5. Air noise spectrum of a centrifugal fan.
1 - Initial fan ($Q = 4100\ m^3/hr$; $H = 131$ mm of water; $\eta = 0.59$); 2 - fan with a mesh on leading and trailing edges of rotor blades ($Q = 4100\ m^3/hr$; $H = 134$ mm of water; $\eta = 0.585$; $D_2 = 325$ mm; $n = 3870$ rpm; $z = 12$ blades).

be obtained by any other known means. The reduction of efficiency may be somewhat compensated by reducing the radial clearance between the trailing edges of the blades and the tongue of the fan volute. In mass-produced ship centrifugal fans, the magnitude of this clearance is $0.1\ D_2$, and represents an attempt to reduce the noise due to the inhomogeneity of the flow. This is manifested in the form of discrete components at the frequency $f = nz/60$ and its multiples (where z is the number of blades, n is the fan rpm).

The use of inclined tongues [4, 5] reduces the clearance in question and almost completely compensates for the worsening of the fan efficiency, caused by the installation of a transition mesh.

A reduction of the noise due to the inhomogeneity of the flow, and consequently a reduction of the tongue inclination angle can be achieved by means of a transition mesh placed on the outer rim of the rotor. This is of great importance, since — when the tongue inclination angles are large — the exit nozzle must be given a more complicated trapezoidal shape instead of the rectangular one, in order to reduce losses in the exit duct.

Conclusion

1. By placing transition meshes at the entrance to the rotor and at the exit from it, in a number of cases it is possible to achieve a reduction in the noise level of centrifugal fans in the entire noise spectrum by as much as 8 - 10 dB.

2. Transition mesh parameters are chosen experimentally. Meshes with various active cross sections are used depending on the spectrum of the fan noise. The greater the fan noise, the smaller the active cross section of the noise should be.

3. In order to compensate for the fan efficiency drop when transition meshes are used, it is useful to reduce the radial clearance between the rotor and the tongue of the volute, and the latter should be tilted.

REFERENCES

1. Novozhilov, S. Ya., Yu. Ya. Petrov and L. M. Yanchenko. Results of the Investigation of Acoustic Flat Profile Lattices. Proceedings of the Sixth All-Union Acoustic Conference, 1968.

2. Ushakov, K. A., A. V. Gusilovskiy and A. R. Bushel. Aerodinamika osevykh ventilatorov i elementy ikh konstruktsii (Aerodynamics of Axial Fans and Elements of their Design). Technical Publishers of Commerce Literature (Gostorgtekhizdat), 1960.

3. Novozhilov, S. Ya., Yu. I. Petrov and G. A. Khoroshev. A Method of Reducing the Aerodynamic Noise Produced by Centrifugal Fans. Patent No. 197918, "Inventions, Industrial Specimens, Brand Names", No. 13, 1967.

4. Khoroshev, G. A. and Yu. I. Petrov. Reduction of the Centrifugal Fan Noise Level. "Sudostroyenie", No. 3, 1965.

5. Petrov, Yu. I. An Investigation of Fan Noise Due to the Inhomogeneity of the Air Flow. Proceedings of the Sixth All-Union Acoustical Conference, 1968.